Simple Thermal Energy Storage Tank for Improving the Energy Efficiency of an Existing Air-conditioning System

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ABSTRACT

This study aims to improve the energy efficiency and reduce the operating time of an existing heating, ventilation, and air-conditioning (HVAC) system by adding a simple thermal energy storage (TES) tank to it. A case study is conducted to evaluate the energy efficiency of the HVAC/TES system using numerical analysis. It is found that the proposed technology saves energy. Increasing the coefficient of performance of the heat-source equipment reduces its energy consumption by 54%. The energy of the primary pumps is reduced by 78% by shortening their operation time. This is achieved by turning off the heat-source equipment and the primary pump while the TES tank discharges heat. The HVAC/TES system is energy efficient. We assess the thermal capacity of a simple TES system and simulate its energy use. Furthermore, we analyze temperature distributions and examine the arrangement of phase-change materials using computational fluid dynamics analysis.

KEYWORDS

Energy saving, Simple thermal energy storage tank, Improved driving efficiency, Latent thermal storage, Existing small- to medium-sized buildings

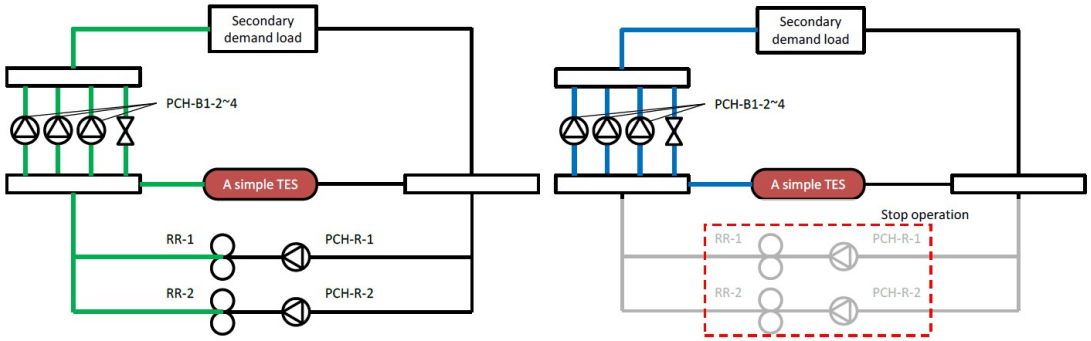
INTRODUCTION

The overall energy consumption in Japan has increased more rapidly in the civilian sector than in any other sector over the past 40 years. Thus, it is necessary to encourage the introduction of energy-saving technologies, especially in the civilian sector. This study focuses on existing small- to medium-sized commercial buildings that have a total floor area $\leq$ 10,000 m^2. Such buildings are likely to have a multisplit-type air-conditioning system, centralized heating, ventilation, and air-conditioning (HVAC) system, or combination of both. Typically, however, the energy management of these systems is inadequate because of budget constraints or insufficient knowledge. Hence, improving the energy efficiency of such systems could save a significant amount of energy. We focus on the centralized HVAC system and propose a technology to improve its operational efficiency. This involves a relatively simple idea of installing small thermal-energy-storage (TES) water tanks in an existing HVAC system to level the heat-load fluctuation due to the charging and discharging heat and achieving a high efficiency of the heat-source equipment by increasing the part-load ratio.

We describe the proposed system in outline and consider how a simple TES tank would work with an existing HVAC system. The TES system comprises a rectangular water tank and phase-change materials (PCMs). It is modeled numerically in relation to the two-dimensional heat transfer between the circulating water and the PCMs. A simulation of the proposed system shows that it can efficiently save energy. Finally, a computational fluid dynamics (CFD) analysis is conducted to modify the PCM configuration for improving the TES efficiency.

OUTLINE OF PROPOSED SYSTEM

A conceptual diagram of the proposed system is shown in Figure 1. The TES tank is added on a bypass pipe and charges heat when the volume of chilled water from the heat-source equipment exceeds that required by the secondary side. Simultaneously, the heat-source equipment improves the operational load factor as heat is charged in the TES tank. Once fully charged, heat is supplied to the secondary side by return chilled water flowing through the tank. Also, the heat-source equipment and the chilled-water pumps can stop the operation. This system is considered acceptable when the heating mode, which is the daily average of the operational load ratio, is low.

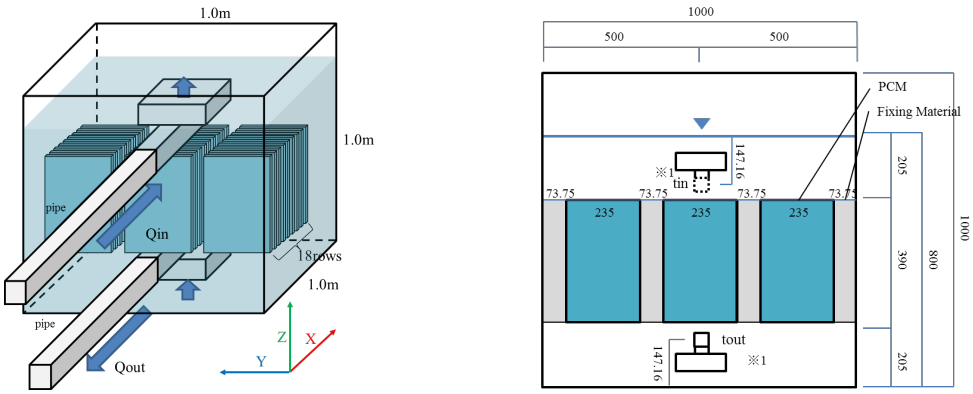


(a) (b)

Figure 1. Charge and discharge modes of proposed system: (a) charge; (b) discharge.

An overview of the TES system is shown in Figure 2. The unit size is 1 m^3, and heating is by temperature stratification; the water level is 80 cm. Diffusers are installed to mitigate the inlet flow velocity (Hinotsu *et al.* 2014). There are 54 PCMs arranged vertically and stabilized with implements.

When the temperature difference is 5 K, the ideal thermal capacity is ~19.8 MJ per unit. Adding PCMs to a thermal water tank could increase its thermal capacity by ~20%.



(a) (b)

Figure 2. Overview of the TES system: (a) perspective view; (b) YZ cross section.

Table 1. Non-dimensional initial depth of complete mixing region (R_0) and Archimedes number (Ar_{in}).

Inlet Flow	Ar_{in}	R_0
10 L/min	39.90	1.34×10^{-2}
100 L/min	0.40	0.22
200 L/min	9.97×10^{-2}	0.50
400 L/min	2.49×10^{-2}	1.15

In Table 1, the relationship between the non-dimensional initial depth of the complete mixing region (R_0) and the Archimedes number (Ar_{in}) is analyzed to define the fundamental TES performance by using an R-value model (Yokota *et al.* 2012). According to Figure 3, $R_0 > 0.5$ when the water flow rate is higher than 200 L/min, which means that it becomes difficult to maintain thermal storage performance. Therefore, the ideal configuration is for the water flow rate per unit to be < 200 L/min.

TES SIMULATION MODEL

An overview of the two-dimensional TES simulation model that is used for the PCM heat transfer is shown in Figure 3 (Hinotsu *et al.* 2014). The model is partitioned length-wise into eight cells. The depth of the model Y is determined by ensuring the same area as that of the real heat-transfer between the PCMs and water. The opening width of the model is Xw. Where there are PCMs, we use Xwp_w and Xwp_p for the width of the water and PCMs, respectively. These values are determined by ensuring the same cubic volume as that of the real system.

In the heat-transfer calculation, only advection is considered between each control volume consisting of water. However, advection, heat conduction in the PCMs, and heat transfer between the water and PCMs are considered in the water/PCM control volume. The time interval of the calculation is determined by the time required for water to circulate in the water/PCM control volume. The PCM temperature is determined by the contained heat quantity, as shown in Figure 4 (the figure is constructed by integrating the heat rate as the standard condition at 40 °C by

differential scanning calorimetry). We ignore heat losses from the tank and the water/PCM heat transfer at planes X = 0 and the max.

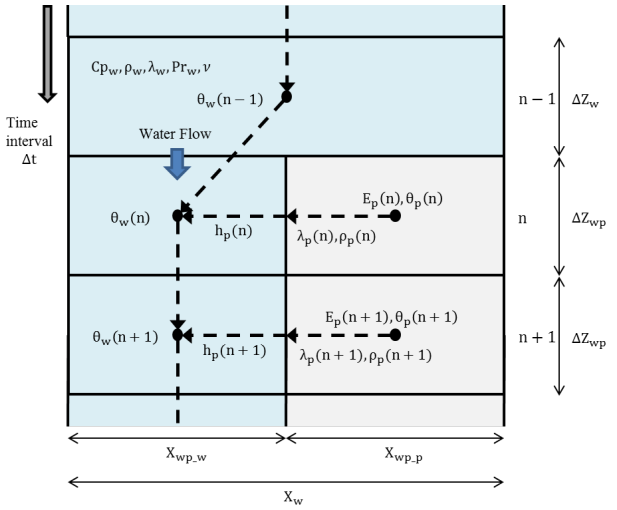


Figure 3. Overview of heat transfer model.

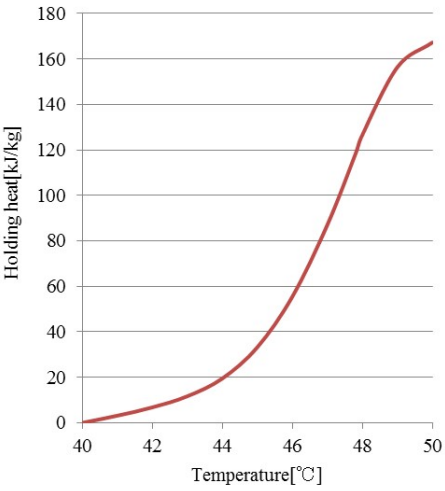


Figure 4. PCM properties.

■ Water-cell $(n = 1,2,7,8)$

$n = 1$

$$\theta_w^1(n) = \theta_w^0(n) + \frac{Cp_w \rho_w Q\left(\theta_{in} - \theta_w^0(n)\right)\Delta t}{Cp_w \rho_w X_w \Delta Z_w Y}$$

$n = 2,7,8$

$$\theta_w^1(n) = \theta_w^0(n) + \frac{Cp_w \rho_w Q\left(\theta_w^0(n-1) - \theta_w^0(n)\right)\Delta t}{Cp_w \rho_w X_w \Delta Z_w Y}$$

■ Water- and PCM-cell $(n = 3 \sim 6)$

Temperature change by advection:

$$\theta_w'^1(n) = \theta_w^0(n) + \frac{Cp_w \rho_w Q\left(\theta_w^0(n-1) - \theta_w^0(n)\right)\Delta t}{Cp_w \rho_w X_{wp_w} \Delta Z_{wp} Y}$$

Temperature change by heat transfer:

$$\theta_w^1(n) = \theta_w'^1(n) - \frac{\left(\theta_p^0(n) - \theta_w'^1(n)\right)\Delta t}{Cp_w \rho_w X_{wp_w}\left(\frac{1}{h^0(n)} + \frac{X_{wp_p}}{2\lambda_p^0(n)}\right)}$$

Heat quantity change of PCM:

$$E_p^1(n) = E_p^0(n) + \frac{-\left(\theta_p^0(n) - \theta_w'^1(n)\right)\Delta t}{\rho_p(n) X_{wp_p}\left(\frac{1}{h^0(n)} + \frac{X_{wp_p}}{2\lambda_p^0(n)}\right)}$$

■ Time interval of calculation:

$$\Delta t = \frac{0.05636475 \times 60 \times 1000}{Q}$$

Cp: Specific heat of water [J/(kg·K)]
E: Holding heat [J/kg]
h: Convective heat transfer coefficient [W/(m²·K)]
Nu_m: Average Nusselt number
Pr: Prandtl number
Q: Inlet flow rate [L/min]
Qh: Heat value [kJ]
Re: Reynolds number
Δt: Time interval [s]
U: Flow rate in the tank [m/s]
ΔX: Width of the cell [m]
Y: Length of the cell [m]
ΔZ: Height of the cell [m]
λ: Thermal conductivity [W/(m·K)]
ν: Kinematic viscosity coefficient of water [m²/s]
θ: Water temperature [℃]
ρ: Density [kg/m³]
0: Present time
1: Next time
in: Input
out: Output
(n): No. of cells
p: PCM
w: Water
wp: Water and PCM

SUMMARY OF SIMULATION

We estimate the effect of the operating efficiency improvement and energy saving of the HVAC/TES system by using a simulation model.

The model building is a ten-story one located in Tokyo. The air-conditioned area is 500 m^2 on each floor, and the total floor area is 5,000 m^2.

The HVAC system equipment is listed in Table 2. The system is a two-pump one installed with air-source heat-pump chillers (basic type) as the heat-source equipment. The scope of the simulation is the primary side of the HVAC system including the secondary pump. We use the flow rate and chilled-water temperature of the return chilled water provided by other system simulation results for the heat load of the secondary side. In this simulation, the chilled-water temperature difference of the secondary side is fixed to 5 K, so the chilled-water flow rate of the return chilled water is calculated based on the heat load.

An HVAC system with the proposed TES system added to an existing bypass pipe is shown in Figure 5. The temperatures at points T4 and T5 shown in the figure determine the switching between heat charging and discharging. The chilled water bypasses through the TES system to avoid increasing the amount of heat-source equipment in operation when the return temperature is too low.

The energy consumption and system-average coefficient of performance (COP) are shown in Figure 6. We conduct a simulation of the heating period from December to March of next year. The energy consumption of the conventional HVAC system is ~37.4 MWh, whereas for the proposed system, it is ~18.2 MWh.

Looking at the breakdown of the results, the energy consumption of the heat-source equipment is reduced by ~54.4%, and that of the primary pump is reduced by ~78%. As shown in Figure 7, the energy consumption of the heat-source equipment is reduced by improving the average COP, which is caused by charging and discharging the TES tank. The primary pump can then be stopped when the tank is discharging.

Table 2. List of HVAC system equipment.

Notation	Name	Specification	Power	Num. of Units
RR-1, 2	Air-cooled heat pump chiller (basic type)	Cooling capacity: 265 kW Rated flow: 760 L/min (7–12°C) COP rating: 2.91	Cooling 91.0 kW	3
		Heating capacity: 300 kW Rated flow: 860 L/min (45–40°C) COP rating: 3.19	Heating 94.0 kW	
PCH-R-1, 2	Primary pump	Rated flow: 860 L/min Lifting height: 250 kPa	5.5 kW	3
PCH-B1-2~4	Secondary pump	Rated flow: 590 L/min Lifting height: 250 kPa	5.5 kW	3

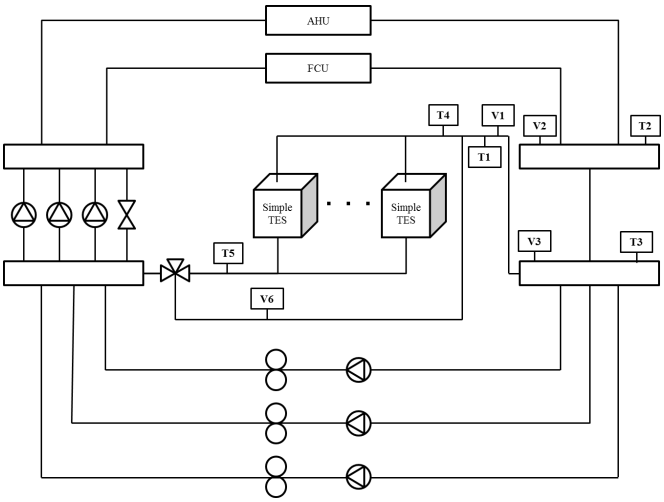


Figure 5. Proposed HVAC system.

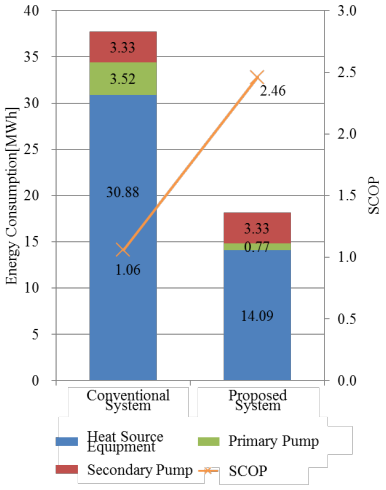


Figure 6. Energy consumption and COP.

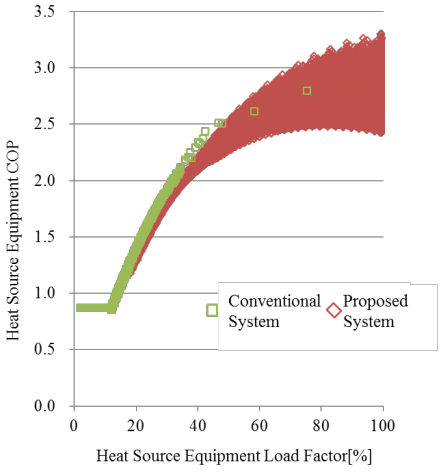


Figure 7. COP and load factor of heat-source equipment.

CFD ANALYSIS

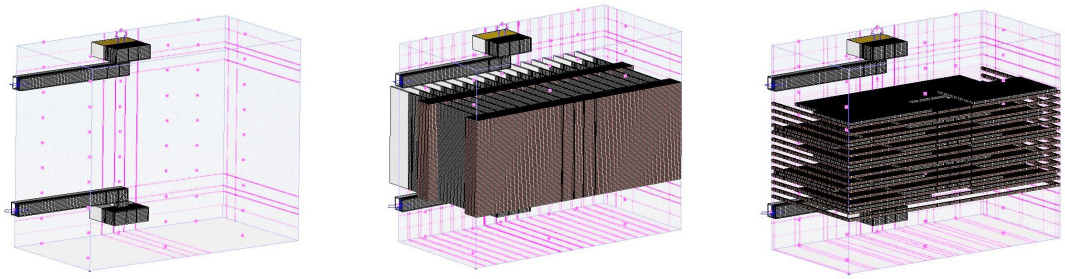
We performed CFD analysis to determine the TES characteristics including the PCM arrangement and to understand the mechanism of temperature stratification. The specifications of the analysis are listed in Table 3. The three-dimensional analysis model is shown in Figure 8: (a) is the model without PCMs; (b) is the model with vertically arranged PCMs (same as Figure 2); and (c) is the model with horizontally arranged PCMs. There are four PCMs on each shelf and 14 overlapping shelves. The temperature difference is 5 K, and the inlet water flow rate is either 100 or 200 L/min. The thermal storage efficiencies of the three cases for the two different water flow rates are listed in Table 4. The thermal storage efficiency is the ratio of the actual storage amount to the ideal one of all of the water in the TES being used at the ideal temperature difference (SHASE, 2006). We evaluate the storage amount which is defined by the thermal stored heat rate during the water changes in the tank completely. According to the table, there is little difference between the model without PCMs and the vertically arranged one. However, there is an improvement

when the PCMs are arranged horizontally. This model increases the thermal storage efficiency even for high inlet flow rates, given the improvement from 100 L/min to 200 L/min.

The relationship between non-dimensional temperature and time when the inlet flow rate is 200 L/min is shown in Figure 9 ((a), (b), and (c) is the same as that in Figure 8). The non-dimensional temperature is the vertical temperature distribution at X = 500 mm and Y = 191.25 mm. According to the figure, the temperature-stratification development in the tank is improved by the horizontal PCM layout because this mitigates the turbulent aspect of the flow in the tank.

Table 3. Specifications of the CFD analysis.

Analysis code		STREAM 11
Fluid		Incompressible
Turbulent flow model		Nonlinear low-Reynolds-number turbulence model
Advection term		Upwind differencing scheme
Boundary conditions	Inflow port	Set a flow rate
	Outflow port	Pressure boundary surface
	Tank wall	Wall: no-slip (Ymax: symmetrical boundary) Water surface: free slip Heat: thermal insulation
No. of mesh points		3,628,278 (no PCMs) 3,899,584 (vertical PCM layout) 3,806,880 (horizontal PCM layout)
Initial temperature		45 ℃
Inlet temperature		50 ℃
Inlet flow		100 or 200 L/min



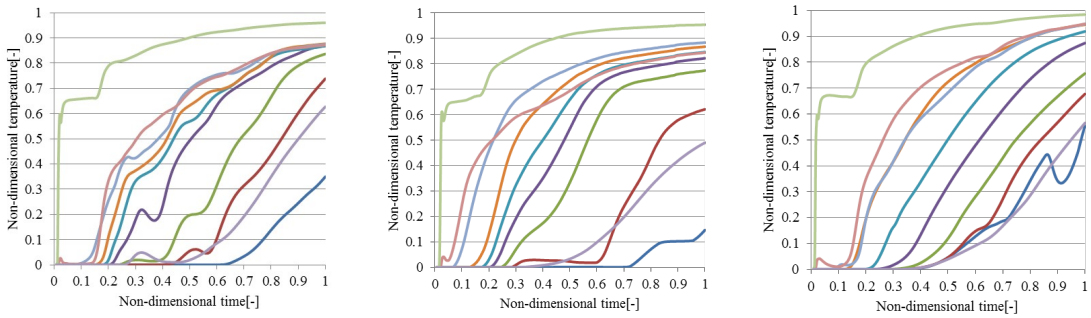
(a) (b) (c)

Figure 8. Three-dimensional analysis model: (a) without PCMs; (b) vertical PCM layout; (c) horizontal PCM layout.

Table 4. Thermal storage efficiency.

PCM layout	Thermal storage efficiency
Without PCM (100 L/min)	88.5

Without PCM (200 L/min)	79.1
Vertical PCM layout (100 L/min)	87.1
Vertical PCM layout (200 L/min)	81.4
Horizontal PCM layout (100 L/min)	89.7
Horizontal PCM layout (200 L/min)	85.3



(a) (b) (c)

Figure 9. Relationship between non-dimensional temperature time: (a) without PCMs; (b) vertical PCM layout; (c) horizontal PCM layout.

CONCLUSIONS

In this paper, the thermal capacity of a simple TES was defined, and energy simulations were conducted to investigate the validity of the proposed system. Furthermore, we examined the arrangement of PCMs by CFD analysis to improve the thermal storage efficiency. We now intend to build an experimental apparatus according to the simulation results, and to improve the precision of the simulation by comparison with the measurement data from the experimental study.

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